Self-Tuning Position Tracking Control of an Electro-Hydraulic Servo System in the Presence of Internal Leakage and Friction

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Self-Tuning Position Tracking Control of an Electro-Hydraulic Servo System in the Presence of Internal Leakage and Friction

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Abstract – *Friction and the internal leakage significantly deteriorates the performance of an electro-hydraulic servo system. Hence, an adaptive scheme could be better choice to handle these effects. Therefore, in this paper a Self-tuning fuzzy PID (SFPID) control scheme is employed for the position tracking performance of an electro-hydraulic servo system. A mathematical model of the system was designed with the consideration of internal leakage and friction inside the system. The internal leakage flow was implemented by introducing several levels and the friction was modeled using LuGre model. The self tuning capability of PID controller was achieved through fuzzy logic theory and the PID controller parameters namely, K_p, K_i and K_d were tuned by selecting appropriate fuzzy rules. The capability of the proposed controller was examined through simulation works in Matlab Simulink and validation was carried out in the real system. The results indicate that the proposed controller successfully improves the positioning performance of the system. The proposed controller can be applied potentially to control the modern equipments positioning applications.*

***Keywords**: Hydraulic Servo, Position Tracking, Self-Tuning Fuzzy Pid, Leakage, Friction*

Nomenclature

A_n	Cross section area (m^2)
a_p	Piston acceleration (m/s^2)
C_d	Discharge coefficient
C_T	Total leakage coefficient
d_u	External disturbance
e	Error trajectory
F_a	Hydraulic actuating force (N)
F_c	Coulomb friction (N)
F_f	Hydraulic friction force (N)
F_s	Static friction (N)
I_{max}	Max. current to valve (A)
K_f	Flow gain
K_{1R},K_{2R}	Flow gain at return port 1 and 2
K_{1S},K_{2S}	Flow gain at supply port 1 and 2
k_f	Leakage coefficient
k_{1R},k_{2R}	Leakage coef. at return port 1 and 2
k_{1S},k_{2S}	Leakage coef. at supply port 1 and 2
k_v	Servo valve gain (m/V)
L	Total stroke of piston (m)
m	Total mass of piston and load (kg)
P_L	Load pressure (Pa)
P_R	Return pressure (Pa)
P_S	Supply pressure (Pa)
P_1, P_2	Pressure in chamber 1 and 2 (Pa)
Q_L	Nominal flow (m^3/s)
Q_{max}	Max. permissible flow (l/min)
Q_s	Internal leakage flow (m^3/s)
Q_1, Q_2	Fluid flow in chamber 1 and 2 (m^3/s)
Q_{1R},Q_{2R}	Return flow at return port 1 and 2 (m^3/s)
Q_{1S},Q_{2S}	Supply flow at supply port 1 and 2 (m^3/s)
u	Input signal to the servo valve (V)
v_p	Piston velocity (m/s)
v_s	Stribeck velocity (m/s)
V_t	Total actuator volume (m^3)
w	Spool valve area gradient (m^2)
x_d	Desired position (m)
x_p	Piston position (m)
x_v	Servo valve spool displacement (m)
x_0	Equivalent orifice opening (m)
z	Average of bristle deflection
σ_0	Bristles stiffness coefficient (N/m)
σ_1	Bristles damping coefficient (N/ms^{-1})
σ_2	Viscous friction (N/ms^{-1})
ρ	Fluid mass density (kg/m^3)

Greek Symbols

ω_v	Servo valve natural frequency (Hz)
β_ε	Effective bulk modulus (Pa)
ζ_v	Servo valve damping ratio
τ_v	Time constant (s)

I. Introduction

Electro-hydraulic servo system is one of the most important equipments which are widely used in modern industrial positioning applications. It has a significant impact due to its abilities in positioning, fast and smooth response, and high power capability. Its applications in positioning involve manufacture assembly lines, flight operations, marine steering system, robotics, etc. In all

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these applications high-accuracy positioning of the system is required. The high-accuracy positioning can be guaranteed when the nonlinearities and uncertainties in the system are compensated. Therefore, an appropriate controller is needed to improve the positioning performance.

Nonlinearities and uncertainties are the common problems in the hydraulic servo systems which include internal leakage and friction. These properties significantly degrade the positioning performance of the system.

Nowadays, the effect of internal leakage and friction on the position tracking performance of an electro-hydraulic servo system has received considerable attention from researchers. Although internal leakage is often ignored by most of the researchers in dynamic analysis and modeling of the system, but in reality, leakage flow between the valve spool and body dominates the orifice flow through the valve [1]. Several observations in [2]-[4] indicate that the internal leakage has a considerable effect on the positioning performance of the system and needs to be considered. Its effect can be categorized in different levels of the leakage orifice opening. Another factor, friction also has a vital influence on system dynamics [5].

Hence it cannot be ignored in the system. It must be estimated earlier to know the model of the friction and to decide a suitable compensator for the friction effect [6]. Studies of the friction effect on the system positioning performance have started by developing a nonlinear observer for Coulomb friction to investigate the existence of static and dynamic friction [7].

The studies were continuing with an invention of an analytic model of dynamic friction which is so called LuGre model. It was proposed to overcome stick-slip motion, pre-sliding displacement and friction lag, because there were several important properties were observed in the system with friction which cannot be explained by static model only [8], [9].

Regarding to the existing controller design for position tracking control of the electro-hydraulic servo system, a simple poles placement design was designed and applied to a linearized model of the system [10]. It was followed by the classic cascaded loops and proportional-integral-derivative (PID) controllers, respectively [11], [12].

An indirect adaptive controller scheme was developed by using pole placement controller [13], [14]. These all aforementioned controllers were designed with the consideration of linear model of the system.

However, the linear controllers contain certain limitations to ensure the tracking accuracy and robustness of the controller, especially for highly nonlinear system like an electro-hydraulic servo system. Therefore, nonlinear controllers which ensure the stability and robustness of the system were developed by employing Lyapunov function.

Lyapunov-based nonlinear controllers were widely used with their main advantage being the lack of restrictions in manipulating nonlinearities in the system [15] including Sliding Mode Control (SMC) and backstepping controller. Robust controllers using SMC were applied to the system to investigate the robustness and position tracking [16]-[18].

Backstepping controllers which constitutes a powerful control strategy for handling the nonlinearities were employed in tracking control of the system [9], [19]. For further improvement, it was combined with adaptive properties [20]-[21].

There were number of efforts were employed by using artificial intelligent techniques. An adaptive control which consists of two Back-propagation networks was designed in [22].

Another approach was the utilization of fuzzy logic theory [23]. Several efforts were carried out by combining the merits of fuzzy and conventional controllers.

A fuzzy inference based PI controller was designed for a better tracking accuracy and smooth control input signals in [8]. Several efforts in development of fuzzy-PID controller and self-tuning fuzzy controller were developed in [24]-[26].

In this study, a SPID controller is developed to compensate the appearance of internal leakage and friction in the electro-hydraulic servo system. SPID is a combination of a classical PID controller and fuzzy logic controller.

The mathematical model of the electro-hydraulic servo system is expressed with the inclusion of the internal leakage and friction model. The influences of the internal leakage and friction to the positioning performance are examined through simulation works in Matlab Simulink environment. Validation is carried out through experimental works in the real system. The internal leakage is classified in several levels, and the friction is modeled using LuGre model.

II. Mathematical Model of Electro-Hydraulic Servo System

Electro-hydraulic servo system equipments involve servo valve, hydraulic cylinder and load attached at the end of the piston as shown in Fig. 1.

The hydraulic cylinder is double-acting hydraulic cylinder with single-rod piston.

When difference between P_1 and P_2 exists, the hydraulic cylinder piston extends or compresses.

The complete mathematical model of the system as shown in Fig. 1 consists of the hydraulic cylinder dynamics including the load environment, and the servo-valve dynamics.

It also describes behaviors of the electro-hydraulic servo system [24], [27]. The mechanical subsystem dynamics of the piston are depending on the load environment

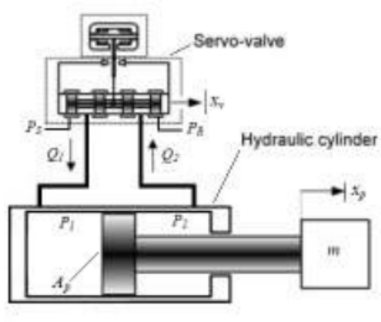

Fig. 1. Electro-hydraulic servo system

The dynamic equations is written as:

$$\dot{x}_p = v_p \tag{1}$$

$$m\dot{v}_p = F_a - F_f - d_u \tag{2}$$

The hydraulic actuating force, F_a and the hydraulic friction force, F_f are commonly derived in the dynamics of servo hydraulic system. The hydraulic actuating force F_a is a nonlinear function of the control input voltage, load environment, cylinder pressure, etc, and it can be represented as:

$$F_a = A_P P_L \tag{3}$$

Hence, equation (2) represents as:

$$m\dot{v}_p = A_p P_L - F_f - d_u \tag{4}$$

In this model:

$$P_L = P_1 - P_2 \tag{5}$$

The differential equations governing the dynamics of the actuator are given. Defining the load pressure to be the pressure across the actuator piston, the derivative of the load pressure P_L, is given by the total load flow through the actuator divided by the fluid capacitance:

$$\frac{V_t}{4\beta_e}\dot{P}_L = Q_L - C_T P_L - A_P v_p \tag{6}$$

Using the equation for hydraulic fluid flow through an orifice, the relationship between spool valve displacement x_v, and the load flow Q_L, is given:

$$Q_L = C_d w x_v \sqrt{\frac{2\left(P_S - sgn(x_v)P_L\right)}{\rho}} + Q_S \tag{7}$$

Therefore, from (4) to (7), the hydraulic dynamics of the actuating force of the cylinder is given by:

$$\dot{P}_L = \\ = -\alpha v_P - \beta P_L + \gamma\left(C_a\sqrt{\frac{2\left(P_S - sgn(x_v)P_L\right)}{\rho}}x_v + Q_S\right) \tag{8}$$

where, $C_a = C_d w$ $\alpha = \frac{4A_P\beta_e}{V_t}$, $\beta = \frac{4C_T\beta_e}{V_t}$, $\gamma = \frac{4\beta_e}{V_t}$.

Spool displacement dynamic equation for of the servo valve x_v, is controlled by an input servo valve u. The corresponding relation can be simplified as:

$$\dot{x}_v = \frac{1}{\tau_v}\left(-x_v + k_v u\right) \tag{9}$$

From equations (1) to (9), if the state variables are selected as $x = \left[x_1, x_2, x_3, x_4\right]^T \equiv \left[x_p, v_p, P_L, x_v\right]^T$, the state equations of the servo hydraulic systems may be written as:

$$\begin{aligned} &\dot{x}_1 = x_2 \\ &\dot{x}_2 = \frac{1}{m}\left(A_P x_3 - F_f - d_u\right) \\ &\dot{x}_3 = -\alpha x_2 - \beta x_3 + \\ &+\gamma\left(C_a\sqrt{\frac{2\left(P_S - sgn(x_{4v})P_L\right)}{\rho}}x_{4v} + Q_S\right) \\ &\dot{x}_4 = -\frac{1}{\tau_v}x_4 - \frac{k_v}{\tau_v}u \end{aligned} \tag{10}$$

III. Internal Leakage and Friction

Q_S and F_f terms as shown in (10) represents the internal leakage and friction behaviors of the system. These behaviors are explained in this section.

III.1. Internal Leakage

Internal leakage is a common phenomenon in the servo valve.

It flows between a servo valve and valve body. In an ideal servo valve, the leakage flows are zero, because it has a perfect geometry.

Practically, the maximum leakage flow occurs at neutral spool position and is only a few percent of the rated flow rate.

The leakage flow decreases rapidly with the valve stroke because of the large overlap between the spool lands and the valve body.

In this study, the mathematically convenient nonlinear servo valve model is performed which accurately captures leakage behaviors of the servo valve over the whole range of spool movement.

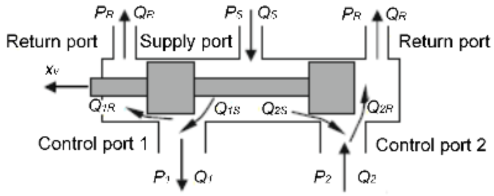

Fig. 2. Servo valve configuration

The leakage behavior is modeled as turbulent flow with a flow area inversely proportional to the overlap between the spool lands and the servo valve orifices [1].

Servo valve configuration consists of two control ports with variable orifices regulate the flow rates, as shown in Fig. 2.

The flow rates through the control ports, and the supply and return ports of the servo valve are expressed as below:

$$Q_1 = Q_{1S} - Q_{1R} \ \ and \ \ Q_2 = Q_{2S} - Q_{2R} \tag{11}$$

$$Q_S = Q_{1S} + Q_{2S} \ \ and \ \ Q_R = Q_{1R} + Q_{2R} \tag{12}$$

The flow rate at the supply and the return side of the port 1 is given by the orifice equation:

$$Q_{1S} = K_{1S}\sqrt{(P_S - P_1)}\left(x_0 + x_v\right) \quad \left(x_v \le 0\right) \tag{13}$$

$$Q_{1R} = K_{1R}\sqrt{(P_1 - P_R)}x_0^2\left(x_0 + k_{1R}x_v\right)^{-1} \quad \left(x_v \ge 0\right) \tag{14}$$

where the parameters x_0 account for leakage flow rate at null ($x_v=0$).

Consider, x_0 is equivalent to a spool displacement that would results in the same amount of flow in a *non-leaking* servo valve as the leakage flow rate in a leaking servo valve with a centered spool [1]. The leakage flow rate is inversely proportional to spool displacement.

From equations (11) to (14) above, we obtain the following relations for control port 1 and port 2 (eqs. (15)-(18)):

$$Q_{1S} = K_{1S}\sqrt{(P_S - P_1)}\begin{cases} \left(x_0 + x_v\right), & \left(x_v \ge 0\right) \\ x_0^2\left(x_0 - k_{1S}x_v\right)^{-1}, & \left(x_v < 0\right) \end{cases}$$

$$Q_{1R} = K_{1R}\sqrt{(P_1 - P_R)}\begin{cases} x_0^2\left(x_0 + k_{1R}x_v\right)^{-1}, & \left(x_v \ge 0\right) \\ \left(x_0 - x_v\right), & \left(x_v < 0\right) \end{cases}$$

$$Q_{2S} = K_{2S}\sqrt{(P_S - P_2)}\begin{cases} x_0^2\left(x_0 + k_{2S}x_v\right)^{-1}, & \left(x_v \ge 0\right) \\ \left(x_0 - x_v\right), & \left(x_v < 0\right) \end{cases}$$

$$Q_{2R} = K_{2R}\sqrt{(P_2 - P_R)}\begin{cases} \left(x_0 + x_v\right), & \left(x_v \ge 0\right) \\ x_0^2\left(x_0 - k_{2R}x_v\right)^{-1}, & \left(x_v < 0\right) \end{cases}$$

Assume the system as a symmetric servo valve with matched control ports [3]:

$$K = K_{1S} = K_{1R} = K_{2S} = K_{2R}$$

$$k = k_{1S} = k_{1R} = k_{2S} = k_{2R}$$

As in [1], the total supply flow Q_S actually represents the internal leakage flow since the control ports are block for an internal leakage test. The internal leakage flow can be expressed as:

$$Q_S = 2K\sqrt{(P_S - P_R)}\left(x_0 + \left|x_v\right|\right)\cdot \\ \cdot\left(1/(1 + f\left(x_v\right)\right)^{-1/2} \tag{19}$$

$$f\left(x_v\right) = \left(1 + \frac{\left|x_v\right|}{x_0}\right)^2\left(1 + k\frac{\left|x_v\right|}{x_0}\right) \\ for \ \ x_v \ge 0 \ \ and \ \ x_v < 0 \tag{20}$$

Servo valve leakage parameters such as K, k and x_0 can be determined by using available manufacturer data of Q_{max} and I_{max} for any types of servo valve [1]. The servo valve leakage flow can be tested by using several level equivalent orifice opening, x_0 from no leak to large leak [3].

III.2. Friction

Friction is a complex and a natural phenomenon which occurs at the physical interface and the tangential reaction force between two surfaces in contact. It can lead to tracking errors, limit cycle oscillation and undesirable stick-slip motion.

Friction also appears in hydraulic cylinder which features strong dry friction effect, due to the tight sealing [5]. Friction is commonly modeled as a discontinuous static mapping between the velocity and the friction force that depends on the velocity's sign which is restricted to Coulomb and viscous friction.

However, some properties cannot be explained by static model only such as stick-slip motion, re-sliding displacement and friction lag, because those properties are dynamic and the fact that the friction does not have an instantaneous response to a change of the velocity [7]. In order to accommodate all properties in static and dynamic friction together, friction force is modeled as LuGre friction model. Velocity-friction diagram is represented as in Fig. 3.

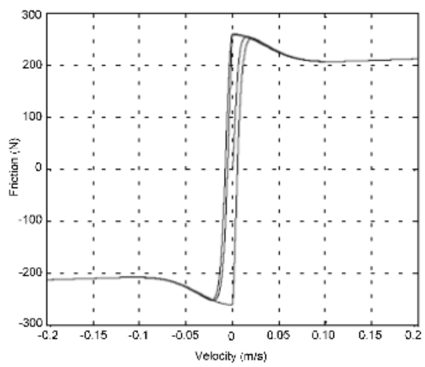

Fig. 3. Friction-velocity description

LuGre friction model is explained as the following equation:

$$F_f = \sigma_0 z + \sigma_1 \dot{z} + \sigma_2 \dot{x}_p \tag{21}$$

$$\dot{z} = \dot{x}_p - \frac{|\dot{x}_p|}{g(\dot{x}_p)} z \tag{22}$$

Servo valve configuration consists of two control ports with variable orifices regulate the flow rates, as shown in Fig. 2. The flow rates through the control ports, and the supply and return ports of the servo valve are expressed as below. The friction internal state represents by variable z describing the average deflections of the bristle between each pair of the contact surface. Friction force parameters σ_0, σ_1, and σ_2 are interpreted as the stiffness of the bristles between two contact surfaces, damping coefficient, and viscous friction coefficient, respectively. Different friction effects are described by nonlinear function $g(\dot{x}_p)$ and can be parameterized to characterize the Stribeck effect:

$$g(\dot{x}_p) = \frac{1}{\sigma_0}\left(F_c + (F_s - F_c)e^{-(\dot{x}_p / v_s)^2}\right) \tag{23}$$

where, F_c, F_s, and v_s respectively are Coulomb friction, viscous friction and Stribeck velocity, respectively. The complete friction model is represented by four static parameters and two dynamic parameters, stiffness coefficient and damping coefficient.

IV. Self-Tuning Fuzzy PID Controller (SPID)

SPID is basically a hybrid controller which combines the conventional PID controller and fuzzy logic principals [28]. The fuzzy logic inference is applied to estimate an appropriate value for the PID controller parameters K_p, K_i and K_d. It was an established method and has been applied in several applications [29]-[32].

IV.1. Conventional PID Controller

PID controller is still the most popular controller, widely used to improve the performance of the hydraulic actuator in industry, because it's easy to operate and very robust. Latest PID controller's structure is quite different from the original one and the implementation is based on a digital design. These digital PID include many algorithms to improve their performance, such as anti wind-up, auto-tuning, adaptive, fuzzy fine-tuning and Neural Networks. However, the basic operations still remain the same.

The performance specifications of the systems such as rise time, overshoot, settling time and error steady state can be improved by tuning the value of parameters K_p, K_i and K_p of the PID controller. Mathematically it is represented as:

$$\begin{cases} y(t) = K_p\left[e(t) + T_d \dfrac{d(e)}{d(t)} + \dfrac{1}{T}\displaystyle\int_0^t e(t)\,d(t)\right] \\ y(t) = \left[K_p e(t) + K_d \dfrac{d(e)}{d(t)} + K_i \displaystyle\int_0^t e(t)\,d(t)\right] \end{cases} \tag{24}$$

where: $K_i = K_p / T_i$, and $K_d = K_p . T_d$.

IV.2. Fuzzy Logic Control Structure

Fuzzy logic controller as shown in Fig. 4, consists of mainly four parts: fuzzification, rule base, inference engine and defuzzification.

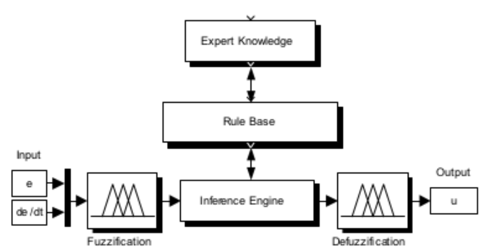

Fig. 4. Fuzzy logic controller block

IV.3. Self-Tuning Fuzzy PID Controller

SPID implies that the three parameters K_p, K_i and K_d of PID controller are tuned by using fuzzy tuner [24]-[26]. Hence, it is necessary to automatically tune the PID parameters for unpredictable parameter variations of a nonlinear system. The structure of the SPID is shown in Fig. 5 where $e(t)$ is the error between desired position set point and the output, $\dot{e}(t)$ is the derivative of error. The PID parameters are tuned using fuzzy inference, which provide a nonlinear mapping from the error and derivation of error to PID parameters.

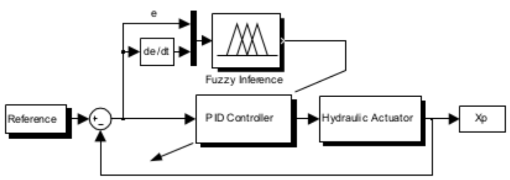

Fig. 5. Structure of SPID

The designed rules are based on the characteristic of the electro-hydraulic actuator and properties of the PID controller. Therefore, the fuzzy reasoning of fuzzy sets of outputs is achieved by aggregation operation of fuzzy sets inputs and the designed fuzzy rules. The aggregation and defuzzification method are max-min and centroid, respectively.

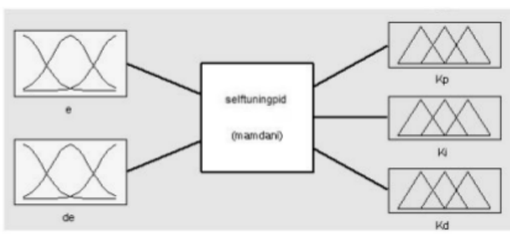

Fig. 6. Fuzzy inference block

For the fuzzy structure, there are two inputs to fuzzy inference: error *e(t)* and derivative of error *de(t)*, and three outputs for each PID controller parameters respectively K'_p, K'_i and K'_d. Mamdani type model is applied in fuzzy inference to obtain the best value for K_p, K_i and K_d. Fuzzy inference block of the controller design is shown in Fig. 6.

Suppose the variable ranges of the parameters K_p, K_i and K_d of PID controller are $\left[K_{p\,min}, K_{p\,max}\right]$, $\left[K_{i\,min}, K_{i\,max}\right]$, and $\left[K_{d\,min}, K_{d\,max}\right]$, respectively. An appropriate range of each parameter is obtained through pre-simulations to get the best initial value of each parameter of the PID controller. This investigation aims to design a feasible rule bases with high inference efficiency. The range of each parameter of the PID:

$$K_p \in [2,\ 1.5]$$
$$K_i \in [0.001,\ 0.0015]$$
$$K_d \in [0.0001,\ 0.00015]$$

Therefore, they can be calibrated over the interval [0, 1] as follows:

$$\begin{cases} K'_p = \dfrac{K_p - K_{i\,min}}{K_{p\,max} - K_{p\,min}} = \dfrac{K_p - 1.5}{2 - 1.5} \\ K'_i = \dfrac{K_i - K_{i\,min}}{K_{i\,max} - K_{i\,min}} = \dfrac{K_i - 0.001}{0.001 - 0.0005} \\ K'_d = \dfrac{K_d - K_{d\,min}}{K_{d\,max} - K_{d\,min}} = \dfrac{K_d - 0.0001}{0.0001 - 0.00005} \end{cases} \tag{25}$$

Hence, from (25), the parameters of the PID controller are obtained by the following formulas:

$$K_p = 0.5K'_p + 1.5$$
$$K_i = 0.0005K'_i + 0.001$$
$$K_d = 0.00005K'_d + 0.0001$$

Membership functions of inputs fuzzy sets are shown in Figs. 7 and 8. The linguistic variable levels are assigned as NB: negative big; NS: negative small; ZE: zero; PS: positive small; PB: positive big. The ranges of these inputs are from -0.17 to 0.17. These ranges are obtained from the absolute values of the system error and its derivative through the gains.

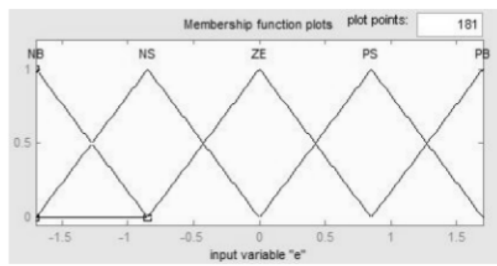

Fig. 7. Membership functions of *e(t)*

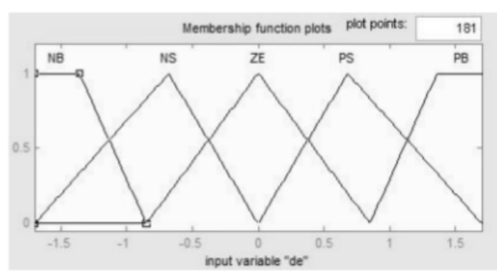

Fig. 8. Membership functions of *de(t)*

Membership functions of outputs K'_p, K'_i and K'_d are expressed in Fig. 9. The linguistic levels of these outputs are assigned as S: small; MS: medium small; M: medium; MB: medium big; B: big, with the ranges from 0 to 1.

In another word, the value of K'_p, K'_i and K'_d should not be less than 0 or more than 1.

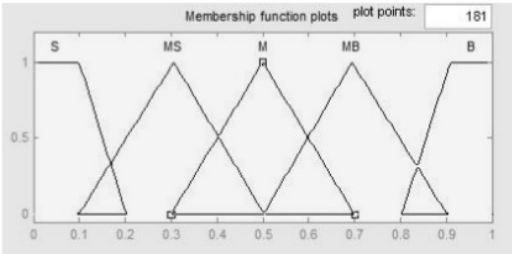

Fig. 9. Membership functions of K'_p, K'_i and K'_d

In this design, the fuzzy rules are developed with the consideration of number of factors such as, system characteristics and specifications to be controlled, type of the controller and from practical experience. The fuzzy rules are shown in Table 1. Rule structure is as follows:

Rule i: If $e(t)$ is A_{1i} and $de(t)$ A_{2i}
then $K'_p = B_i$ and $K'_i = C_i$ and $K'_d = D_i$

where, i= 1, 2, 3,..., n, and n is number of rules. From the table, since we have 5 variables for both input and output, hence, a maximum of 25 fuzzy rules can be used as here too.

TABLE I
RULES OF FUZZY INFERENCE

de\e	NB	NS	ZE	PS	PB
NB	S	S	MS	MS	M
NS	S	MS	MS	M	MB
ZE	MS	MS	M	MB	MB
PS	MS	M	MB	MB	B
PB	M	MB	MB	B	B

V. System Parameters and Setup

The performance of the proposed controller is examined through simulation and experimental works. Parameters of the electro-hydraulic servo system which are used in this study for the simulation and experimental works are obtained from the manufacturer datasheet of the existing system in the Laboratory. The parameters are as can be seen in Table II.

The system which is modeled and controlled in this study is composed of double acting hydraulic with single-rod piston driven by a direct servo valve Bosch Rexroth 4WREE6, and 40 lpm flow rate at 70 Bar.

Dimension of hydraulic cylinder are 63 mm/30 mm/300 mm. 300 mm draw wire sensor is used to measure the piston position. 100 bar pressure transducers are attached to measure the pressure into and from the cylinder.

DAQ card NI PCI 6221 is employed as interface between Matlab's programs in PC with the system test bed.

A set of load environment is attached at end of the piston. Experimental setup of the system test bed is as shown in Fig. 10.

TABLE II
PARAMETERS OF THE SYSTEM

Cylinder parameters		
P_S	Supply pressure (Pa)	$0.7\text{x}10^7$
P_R	Supply return (Pa)	0
V_t	Total actuator volume (m^3)	$0.89\text{x}10^{-3}$
A_p	Actuator ram area (m^2)	$2.97\text{x}10^{-3}$
L	Total stroke of piston (m)	0.3
m	Total mass of piston- load (kg)	18
β_e	Effective bulk modulus (Pa)	$1\text{x}10^9$
ρ	Fluid mass density (kg/m^2)	850
Servo valve parameters		
C_d	Discharge coefficient	0.6
w	Spool valve area gradient (m^2)	0.02
k_v	Servo valve spool gain (m/V)	$1.27\text{x}10^{-5}$
I_{max}	Max. rate current to servo (A)	2
Q_{max}	Max. permissible flow (l/min)	80
Leakage parameters		
x_0	Equivalent orifice opening	0 to $12\text{x}10^{-4}$
k	Leakage coefficient	0.3
K	Flow gain	$1.42\text{x}10^{-5}$
Friction parameters		
F_s	Static friction (N)	260
F_c	Coulomb friction (N)	200
σ_0	Bristles stiffness coeff. (N/m)	$12\text{x}10^5$
σ_1	Bristles damping coeff. (N/ms^{-1})	300
σ_2	Viscous friction (N/ms^{-1})	60
v_s	Stribeck velocity (m/s)	0.1

The proposed controller is applied to supply a control input signal u into the servo valve with voltage unit. The control signal is employed to control the position of the spool of the servo valve. The spool position influences the nominal flow which flows into the cylinder Q_1 and from the cylinder Q_2. A change in the nominal flow results a change in the volume and pressure in each chamber of the cylinder. Finally, it can affect the piston position of the system. The positions of the spool and piston are measured in meter unit.

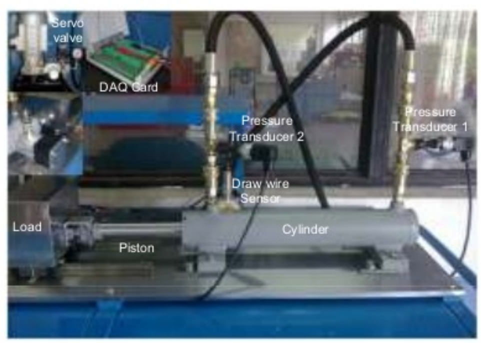

Fig. 10. Experimental set-up

SPID subsystem block is shown in Fig. 11. It consists of fuzzy logic controller and PID controller blocks with the proposed modifications. The values of K_p, K_i and K_d

are tuned based on the error e and the changes of error $\dot{e}(t)$. It refers to the formula in (25) which is used to determine the values of K'_p, K'_i and K'_d from the fuzzy logic controller block. The values of K'_p, K'_i and K'_d are multiply with the range between maximum values and minimum values of K_p, K_i and K_d. The multiplication results are then added by initial values of K_p, K_i and K_d to obtain the final values of the proposed controller parameters.

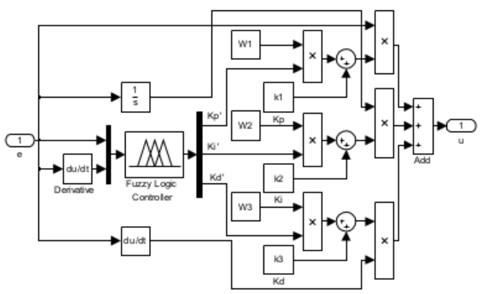

Fig. 11. Simulink block of fuzzy PID regulator

Simulation and experimental works on the complete system are carried out in the following steps. Four types of simulation scenarios are discussed here and these are categorized in steps 1-4. Experimental work is categorized as Step 5. All these steps are performed to examine the proposed controller design ability to overcome the degradation of the positioning performance and to satisfy the objectives of the study:

Step 1: In this step the effect of internal leakage and friction on the positioning performance of the system are observed, by injecting 0.05 m step input signal into the system without controller.

Step 2: Capability of the SPID and the conventional PID controller are examined and compared for the positioning performance of the system. Five (5) different levels of leakage are used for the controller investigation. These are obtained by changing the value of equivalent orifice opening x_0. Level 1 to 5 are given 0.0 m, 1×10^{-5} m, 2×10^{-5} m, 4×10^{-5} m, and 8×10^{-5} m, respectively. Step input signals with 0.05 m are employed in this examination.

Step 3: Sensitivity of both controllers are tested and compared in compensating the internal leakage and the friction, respectively. The internal leakage which is represented by a medium level (level 3) exists in the system during the investigation. Step input signal with 0.05 m is used in both investigations.

Step 4: Comparison of the SPID over the conventional PID controller in handling the appearance of internal leakage and friction is shown. Both controllers are tested separately for the system with internal leakage, friction, and combination of both. The internal leakage is set as level 3. Step input signal with 0.05 m is applied in all conditions, but sine wave signal is only employed to the system with both internal leakage and friction.

Step 5: Experiments are carried out to examine the proposed controller in the real system when the internal leakage and friction are present in the system. Two types of reference signals are utilized here, namely step and sine wave input.

VI. Results and Discussion

VI.1. Simulation

Responses of internal leakage and friction effect on the positioning performance of the electro-hydraulic servo system are shown in Fig. 12.

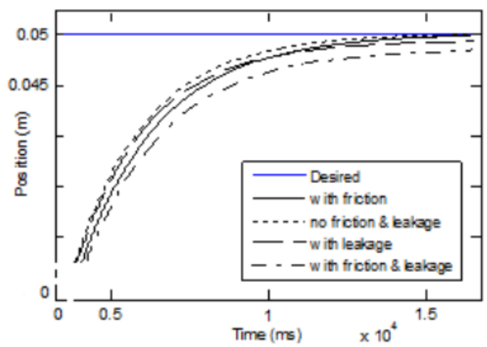

Fig. 12. Effect of internal leakage and friction

From Fig. 12, when the system without controller is examined using the step input, internal leakage have a dominant effect to that of friction for the positioning performance of the system.

The performance will tend to degrade, when both internal leakage and friction exist in the system.

Figures 13–15 show the position responses of the system for five (5) different levels of internal leakage. As can be seen in Fig. 13, when level of the leakage changed from a lower level to a higher level, the positioning performance of the system increases significantly toward the desired output, because the maximum leakage flows in the servo valve occurs at neutral spool position. The leakage flow decreases rapidly with the valve stroke.

Figs. 14 and 15 show the performance of PID controller and SPID controller in compensating the effect of the internal leakage.

Both controllers are capable to compensate the appearance of the internal leakage compare to the system without controller as shown in Fig. 13.

However, SPID controller is better than PID controller in handling the presence of the internal leakage in the system.

The SPID can handle the error dynamics better than conventional PID controller.

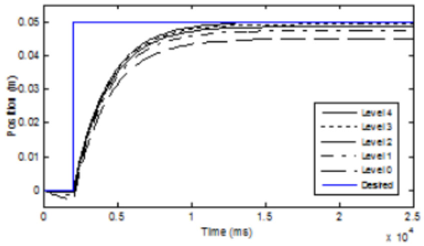

Fig. 13. Responses with several levels of leakage without controller

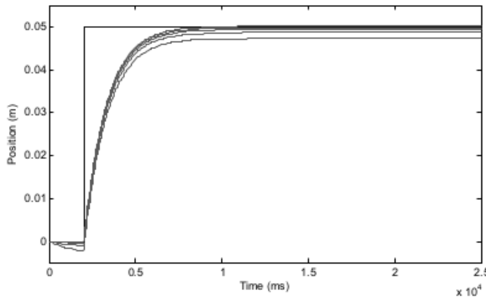

Fig. 14. Responses with several levels of leakage using PID controller

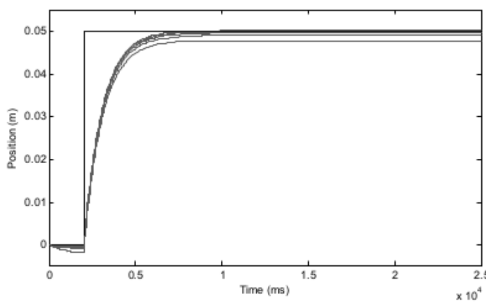

Fig. 15. Responses with several levels of leakage using fuzzy-PID controller

Figs. 16–19 depict the comparison of the both controllers.

This comparison is evaluated in handling the presence of internal leakage, friction, and combination of internal leakage and friction, respectively.

For a medium level of the internal leakage, Fig. 16 indicates that the system which utilizes the SPID offers a better tracking performance for positioning to that of conventional PID controller.

Moreover, the system having a SPID controller shows a better response in compensating the presence of friction as shown in Fig. 17.

Figs. 18-19 show the performance of both controllers with the consideration of internal leakage and friction against two types of reference signals. For each reference signal, SPID offers better performance than the PID controller. A more accurate tracking can be observed for the SPID controller.

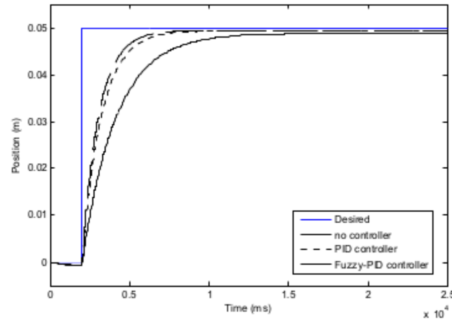

Fig. 16. Responses with internal leakage using step input

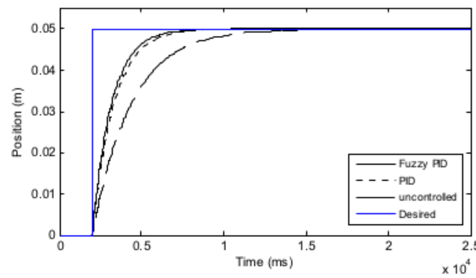

Fig. 17. Responses with friction using step input

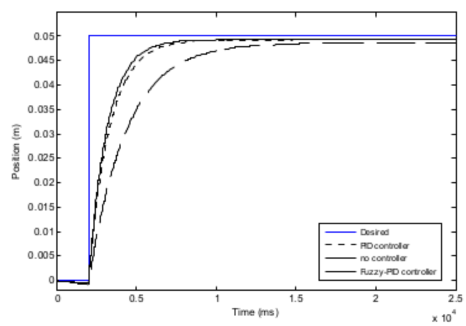

Fig. 18. Responses with friction and leakage using step input

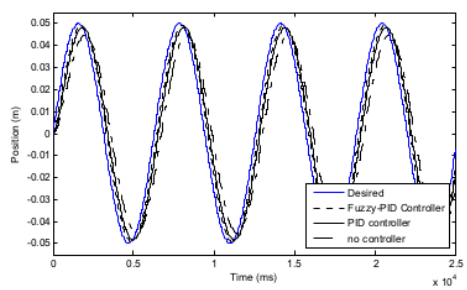

Fig. 19. Responses with friction and leakage using sine wave input

Hence, in general SPID controller design is better than conventional PID controller to control the position tracking of the system in the presence of internal leakage and friction.

VI.2. Experimental

Based on the system test bed as shown in Fig. 10, experimental works on the real system are performed with the same settings and parameters as carried out in the simulation.

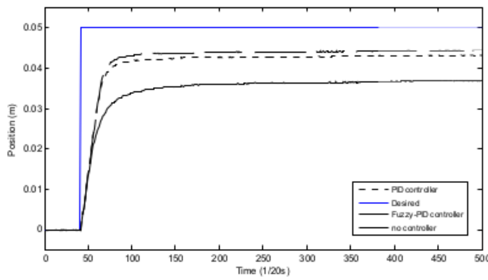

Fig. 20. Responses with friction and leakage using step input

From Figs. 20 and 21, it can be observed that the system with SPID shows better performance than the conventional PID controller to control the position of the electro-hydraulic system. However, the actual position of the piston cannot be track to the desired position accurately as was in the simulation responses. This is due to the behavior of the electro-hydraulic servo system, which is highly nonlinear, uncertain and time-varying. Moreover, internal leakage and friction are only a few parts of nonlinearities and uncertainties which appear in an electro-hydraulic servo system. For these reasons, we need to design a suitable nonlinear controller and a better estimation for the internal leakage and friction.

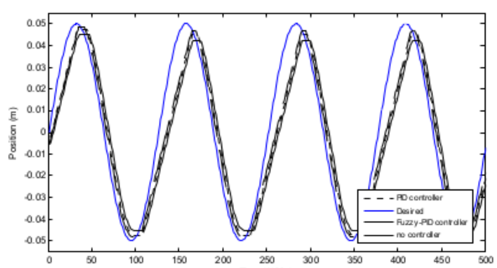

Fig. 21. Responses with friction and leakage using sine wave input

VII. Conclusion

Modeling for internal leakage and friction was discussed for an electro-hydraulic servo system. The proposed SPID was successfully developed in simulations and experiments. A self-tuning fuzzy regulator was applied to tune the values of K_p, K_i and K_d of the PID controller. Capabilities of controllers were investigated with the consideration of internal leakage and friction model. Results reveal that the appearance of the internal leakage and friction significantly degrades the accuracy of the tracking position. SPID controller offers better results compare to conventional PID controller in compensating the presence of the internal leakage and friction.

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